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Investigation of ammonia/water hybrid absorption/compression heat pumps for heat supply temperatures above 100 °C

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ABSTRACT

The hybrid absorption/compression heat pump (HACHP) using ammonia-water as working fluid is a promising technology for development of a high temperature industrial heat pump. This is due to two properties inherent to the use of zeotropic mixtures: non-isothermal phase change and reduced vapour pressures. Using standard refrigeration components (28 bar) HACHP up to 100 °C are commercially available. Components developed for high pressure NH_3 (52 bar) and transcritical CO_2 (140 bar) increase the limiting allowable pressures. It is therefore relevant to evaluate the feasible supply temperatures using these components. A technically and economically feasible solution is defined as one that satisfies constraints on the coefficient of performance (COP), low and high pressure, compressor discharge temperature and volumetric heat capacity. The ammonia mass fraction of the rich solution and the liquid circulation ratio both influence these constraints. The paper investigates feasible combinations of these parameters. A numerical HACHP model is developed in Engineering Equation Solver (EES). The constrained parameters are evaluated for a range of combinations for systems with supply temperatures of 100 °C, 125 °C, 150 °C and 175 °C. Results show that standard components are applicable up to 100 °C, equipment for high pressure NH_3 up to 125 °C, and equipment for transcritical CO_2 up to 175 °C.

1. INTRODUCTION

The hybrid absorption/compression heap pump (HACHP), or vapour compression heat pump with solution circuit is based on the Osenbrück cycle (Osenbrück, 1895). In the Osenbrück cycle the condensation and evaporation processes are exchanged with absorption and desorption processes. It thus uses zeotropic mixtures as the heat pump working fluid, typically ammonia/water.

The first theoretical study of the HACHP was performed by Altenkirch (1950) and described the advantage of the HACHP heat pump with the non-isothermal process of absorption/desorption compared to the isothermal process of condensation/evaporation in a conventional vapour compression heat pump (VCHP). Thereby the cycle approaches the Lorenz cycle (Lorenz, 1894) which can result in an increased COP due to the reduction of entropy generation driven by heat transfer over a finite temperature difference. The efficiency advantage of the HACHP over the VCHP requires the temperature change (glide) in the heat sink and heat source to be greater than 10 K (Hultén and Berntsson, 2002). The advantage remains even if economic considerations are included in the comparison (Hultén and Berntsson, 1999). This makes the HACHP a relevant technology for industrial heat supply and waste heat recovery as these processes often require large sink/source temperature glides.

A further advantage of using a zeotropic mixture as working fluid is the reduction of vapour pressure compared to the vapour pressure of the pure volatile component. This implies that the HACHP can achieve higher supply temperatures than a VCHP at the same working pressure. Therefore the HACHP is of specific interest for high temperature applications. Brunin *et al.* (1997) showed that it is technically and economically feasible to use the HACHP up to a heat supply temperature of 140 °C, this however is based on a high pressure constraint of 20 bar corresponding to the limitations of standard refrigeration components at the time of the study. In the meantime new compressor types such as high pressure NH_3 (52 bar) and transcritical CO_2 (140 bar) have become commercially available. It is therefore of interest to evaluate how the application of these components changes the working domain of the HACHP.

One design constraint that is not discussed in the study by Brunin *et al.* (1997) is compressor discharge temperature. However, most compressor manufacturers require this to be lower than 180 °C (Nekså *et al.* 1998). This is mainly due the thermal stability of the lubricating oil and the thermal stress of the materials surrounding the compressor discharge line. Changing the lubricant from a mineral oil to synthetic oil could relax the constraint due to thermal stability. This however requires that a synthetic oil that meets the requirements of miscibility etc. is identified. Small adjustments to the gasket materials and alike could also make the compressor durable at higher discharge temperatures. It is assumed to be a realistic estimate that compressor discharge temperatures up to 250 °C can be sustained with minor adjustments.

To evaluate the working domain of the HACHP using the recently developed high pressure equipment a set of design constraints are defined. A solution that satisfies this set of constraints will constitute an economically and technically feasible solution. The technical limitations are: the high pressure, governed by the choice of compressor technology, the low pressure, set to eliminate entrainment of air, and the compressor discharge temperature, as discussed above. The economic constraints are: the coefficient of performance (COP) and the volumetric heat capacity (VHC), calculated as the ratio of the compressor displacement volume to the heat output of the HACHP (Brunin *et al.* 1997).

Two design parameters highly influence all the design constraints. These are the rich ammonia mass fraction x_r and the circulation ratio f, defined as the ratio between the mass flow rate of the rich solution and the lean solution. The combination of these two govern the system pressure and thereby the VHC. Also the slope of the absorption/desorption curve and thereby the performance (COP) is influenced by these parameters. These relations have been addressed with different approaches by Stokar (1987), Åhlby *et al.* (1991), Rane and Radermacher (1992), Rane *et al.* (1993), Hultén and Berntsson (1999, 2002) and latest by Zamjlrescu (2009) .

The present study investigates the set of feasible combination of x_r and f at heat supply temperatures of 100 °C, 125 °C, 150 °C and 175 °C. This will be evaluated for all three mentioned types of refrigeration components.

2. MODELLING & ANALYSIS OF THE HYBRID HEAT PUMP

The process diagram of the evaluated HACHP may be seen in Figure 1. The working principle of the HACHP is as follows. At the outlet of the desorber the working fluid is a liquid vapour mixture. The liquid phase will have a lean concentration of ammonia, while the vapour phase consists primarily of ammonia (at low pressures the water vapour content can be substantial). The bulk ammonia concentration of the stream exiting the desorber is the rich ammonia mass fraction.

Before elevating the pressure the vapour and liquid phases must be separated such that liquid does not enter the compressor. This is done in the liquid/vapour separator placed after the desorber. Here the liquid settles in the bottom of the tank while vapour stays at the top. The vapour is then drawn to the compressor in which the pressure and temperature are increased. Next the vapour is delivered to the gas cooler in which the vapour temperature is reduced while releasing heat to the sink. The liquid is drawn to the pump where the pressure is elevated to the high pressure. As the liquid is incompressible the temperature increase over the pump is small. Therefore the liquid is heated in the internal heat exchanger. This reduces the entropy generation when mixing the liquid and vapour at the high pressure.

Prior to the absorber the vapour and liquid phases are mixed in an adiabatic process which results in a vapour liquid mixture in thermal, mechanical and chemical equilibrium. This would usually not be a separate component, but is an

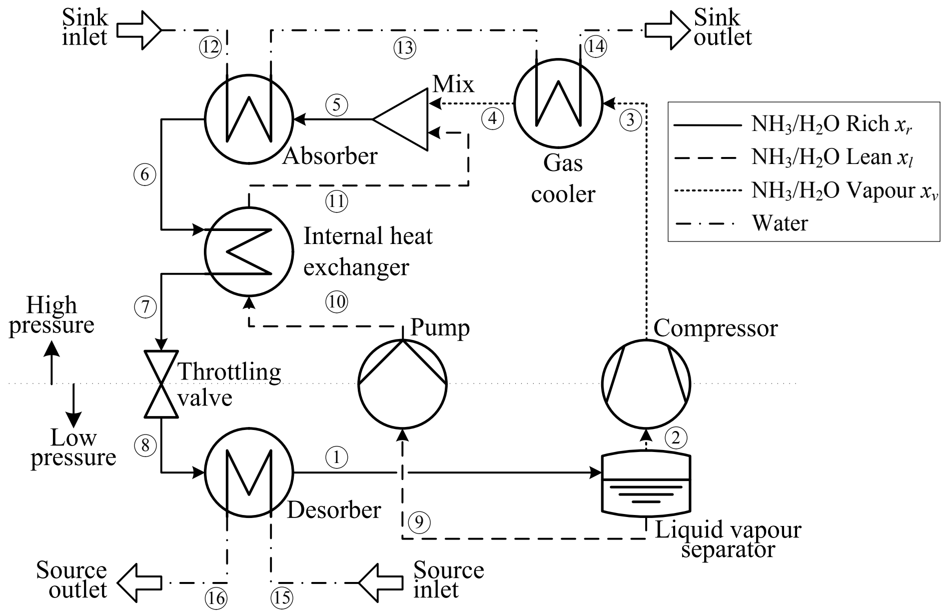

Figure 1. Schematic of the hybrid absorption/compression heat pump.

adiabatic absorption process that takes place in the first part of the absorber. In a thermodynamic model both treatments will yield the same result. When the equilibrium state is reached the diabatic absorption process begins. Here the ammonia is absorbed in the lean liquid phase while releasing heat to the sink

At the absorber outlet a saturated liquid rich in ammonia is delivered. This is sub-cooled in the internal heat exchanger. The sub-cooled rich ammonia solution is throttled to the low pressure resulting in a low temperature liquid vapour mixture. This is feed to the desorber in which vapour is generated by supplying heat from the source.

The process described above is sketched in the temperature – heat load diagram shown in Figure 2. Here the temperature lift, ΔT_{lift}, is defined as the difference between the sink outlet temperature (heat supply temperature) and the source inlet temperature. The temperature glide, $\Delta T_{\mathrm{glide}}$, is defined as the temperature difference between the inlet and outlet of the sink and source respectively. Further it may be seen that the profiles of the absorption and desorption processes are non-linear. This has been described in detail by Itard and Machielsen (1994). In Figure 2 these are depicted as convex curves. Dependent on ammonia mass fraction and circulation ratios these profiles could also exhibit a concave curve or have a convex and a concave part. This is well described in Zheng *et al.* (2013). This means that when modelling the HACHP it is not sufficient to ensure positive temperature difference at the inlet and outlet of the absorber and desorber. To ensure a feasible profile it is necessary to verify that there is a positive temperature difference over the entire heat transfer process.

2.1 Thermodynamic modelling of hybrid absorption/compression heat pump

A numerical model of a HACHP has been developed in Engineering Equation Solver (EES) (Klein, 2013). The thermodynamic properties of the ammonia water mixture is calculated using EES's external routine "NH3H2O" which is based on equations of state developed by Ibrahim and Klein (1993). Each component is modelled based on a steady state mass and energy balance. Further the model ensures that the second law is fulfilled in all components.

Heat losses from components and pressure losses are neglected. The rich ammonia mass fraction x_r and circulation ratio f are inputs to the model. The rich ammonia mass fraction is present in state 5-8 and 1. The circulation ratio is defined as the ratio between the mass flow rate of the rich solution $\dot{m}_r$ and the lean solution $\dot{m}_l$, $f = \frac{\dot{m}_l}{\dot{m}_r}$. Hence the circulation ratio is directly linked to the vapour quality in state 1 exiting the desorber, such that: $q_1 = 1 - f$. From

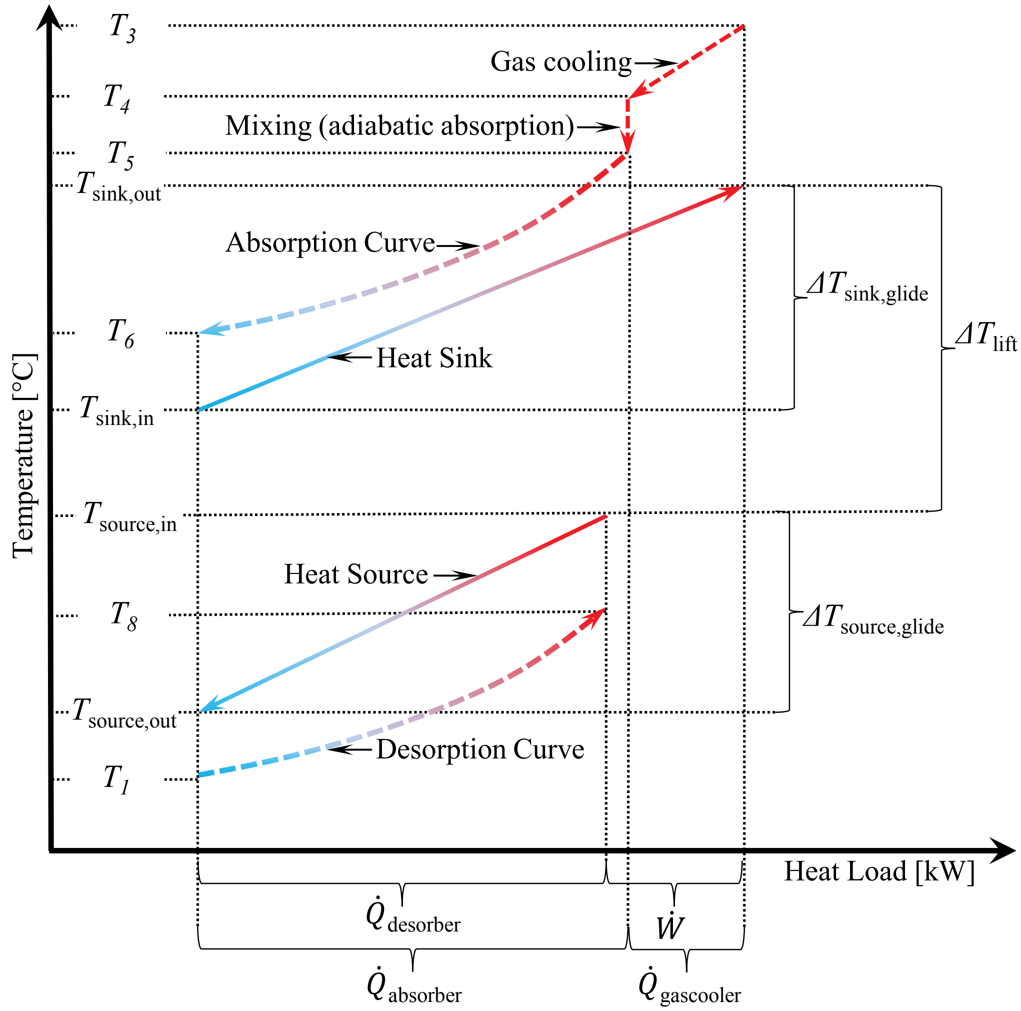

Figure 2. Principle temperature – heat load diagram for the hybrid absorption compression heat pump

this is can be concluded that if the circulation ratio is 0 then the HACHP is essentially a VCHP with a zeotropic working fluid.

As pressure loss and heat losses are neglected the temperature and pressure in the liquid/vapour separator are the same as that exiting the desorber. It is assumed that the vapour and liquid exiting the desorber are saturated, $q_2 = 1$ and $q_9 = 0$. The vapour and lean ammonia mass fraction, x_v and x_l, is then determined.

The processes in the compressor and pump are modelled as adiabatic with given isentropic efficiencies, η_{is}. The transferred heat in the internal heat exchanger and gas cooler is calculated based on given values of heat exchanger effectiveness, ε. The mixing or adiabatic absorption process found prior to the absorber is modelled only by a mass and energy balance. Thus state 5 is the equilibrium state attained when mixing stream 4 and 11. The state exiting the desorber is assumed to be saturated, $q_6 = 0$. The forced adiabatic expansion in the throttling valve is assumed to be isenthalpic, $h_7 = h_8$.

The high and low pressures, p_H and p_L, are determined to satisfy given values of pinch point temperature difference, ΔT_{pp}, in the absorber and desorber. To account for the non-linearity of the absorption and desorption curve these components are discretised in heat load, giving the specific enthalpy of both the sink/source and ammonia/water mixture at each step. Assuming constant pressure and constant bulk ammonia mass fraction the equilibrium

Table 1: Component inputs to the thermodynamic model

	$\Delta T_{pp,abs}$	$\Delta T_{pp,des}$	$\eta_{is,comp}$	$\eta_{vol,comp}$	η_e	$\eta_{is,pump}$	ε_{ihex}	ε_{gc}
Input value	5K	5K	0.75	0.95	0.90	0.75	0.85	0.85

temperatures and temperature differences are attained at each step. The pinch point temperature difference is defined as the minimum of these temperature differences. 30 steps are used in this model for both the absorber and desorber. The COP of the HACHP is defined as given in equation (1). Here $\dot{W}_{\text{comp}}$ and $\dot{W}_{\text{pump}}$ are the work calculated based on the given isentropic efficiencies. The efficiency of the electric motors driving the pump and compressor is account for by the electric efficiency, η_e.

$$\text{COP} = \frac{\dot{Q}_{\text{abs}}+\dot{Q}_{\text{gc}}}{\dot{W}_{\text{comp}}/\eta_e+\dot{W}_{\text{pump}}/\eta_e} \tag{1}$$

The displacement volume of the compressor, $\dot{V}_{\text{dis}}$, is found by equation (2), here $\dot{V}_{\text{suc}}$ is the suction line volume flow rate, $\eta_{\text{vol,comp}}$ is the volumetric efficiency of the compressor and v_2 is the specific volume of state 2.

$$\dot{V}_{\text{dis}} = \frac{\dot{V}_{\text{suc}}}{\eta_{\text{vol,comp}}} = \frac{\dot{m}_2 v_2}{\eta_{\text{vol,comp}}} \tag{2}$$

The VHC is calculated as the ratio between the heat output of the HACHP and the displacement volume of the compressor, see equation (3). VHC is thus a measure of the size of compressor needed to deliver a certain heat load.

$$\text{VHC} = \frac{\dot{Q}_{\text{abs}}+\dot{Q}_{\text{gc}}}{\dot{V}_{\text{dis}}} \tag{3}$$

The pinch point temperature difference of absorber and desorber, isentropic, volumetric and electrical efficiency of the compressor and pump and the effectiveness of the internal heat exchanger and gas cooler are fixed parameters. The applied values are listed in Table 1.

2.2 Design constraints

The design constraints are listed in Table 2. The constraint on COP and VHC ensures the economic feasibility of the heat pump as a high COP ensures a low running cost and a high VHC ensure a low investment cost. The applied values for the COP and VHC constraint are set in agreement with those presented by Brunin *et al.* (1997). Here the constraint on the low pressure is also presented. This is set to 1 bar to prevent sub-atmospheric pressures and remove the risk of air entrainment. The high pressure constraint for the different compressor technologies are summarized by Ommen *et al.* (2011). Nekså *et al.* (1998) state that compressor discharge temperatures up to 180 °C should be possible without degeneration of lubricant while Ommen *et al.* (2011) state this could be as low as 160 °C. The value for the unmodified compressors is set to 170 °C in this study. For the modified transcritical CO_2, 250 °C is chosen as discussed earlier.

3. RESULTS

The design of the HACHP has two more degrees of freedom compared the VCHP. The design of a VCHP can be determined using only the component inputs from Table 1 and external operating condition: $\Delta T_{\text{sink,glide}}$, ΔT_{lift}, $T_{\text{sink,out}}$, $\dot{m}_{\text{sink}}$ and $\dot{m}_{\text{source}}$. Thus all the constrained parameters can be determined based on the inputs. For the HACHP the circulation ratio f and the ammonia mass fraction x_r must also be given. It is therefore of relevance to investigate

Table 2: Design constraints for standard refrigeration, high pressure NH_3 and trans-critical CO_2 components

	p_H	T_H	COP	VHC	p_L
Standard ref.	28 bar	170 °C	4 -	2000 kJ/m^3	1 bar
High pressure NH_3	52 bar	170 °C	4 -	2000 kJ/m^3	1 bar
Transcritical CO_2	140 bar	250 °C	4 -	2000 kJ/m^3	1 bar

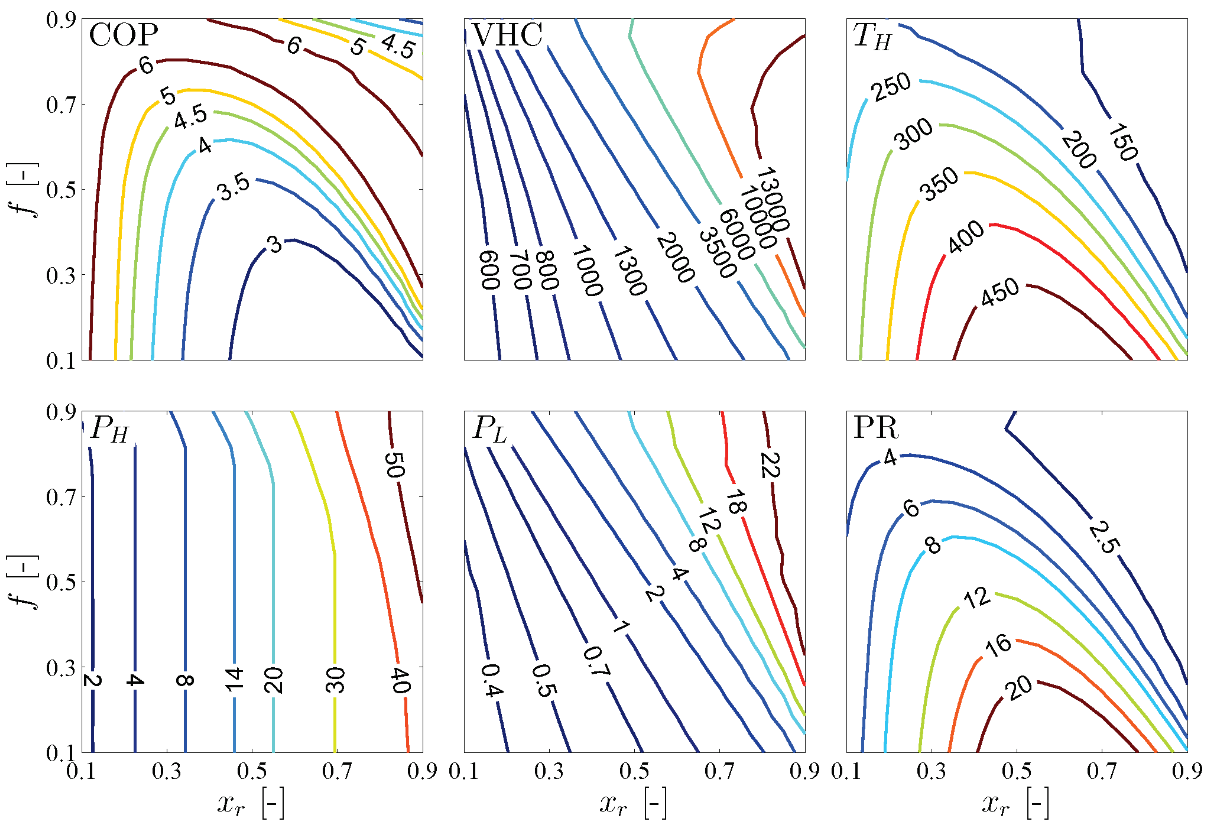

Figure 3. COP, VHC, compressor discharge temperature T_H, high pressure P_H, low pressure P_L and pressure ratio *PR*, as a function of the circulation ratio *f* and the ammonia mass fraction x_r. Heat supply temperature $T_{sink,out}$=100 °C, sink glide $\Delta T_{sink,glide}$=20 °C, temperature lift ΔT_{lift}=25 °C

how the choice of these inputs influence the design of the HACHP.

3.1 Influence of x_r and *f*

Figure 3 shows the COP, VHC, compressor discharge temperature T_H, high pressure p_H, low pressure p_L and pressure ratio (PR) for the HACHP. The component inputs listed in Table 1 and design operating point are held constant. The operating conditions are $T_{sink,out}$=100 °C, $\Delta T_{sink,glide}$=20 K, ΔT_{lift}=25 K, $\dot{m}_{\text{sink}} = 1$ kg/s and $\dot{m}_{\text{source}}$=2 kg/s

When comparing the contours of the high and low pressure it becomes evident that the circulation ratio has a greater influence on the low pressure than on the high pressure. The circulation ratio only influences the high pressure at high ammonia mass fractions and high circulation ratios. The low pressure is governed by both parameters such that an increase in ammonia mass fraction and an increase in circulation ratio will increase the low pressure. Thus the lowest low pressure is at x_r=0.1, *f*=0.1 and the highest low pressure is at x_r=0.9, *f*=0.9. From this it is apparent that a set of combinations induce a significant increase in the system pressure ratio. These are combinations with low circulation ratio and ammonia mass fractions around 0.5. This further induces a large increase in compressor discharge temperature varying from 150 °C for the low pressure ratio combinations to 450 °C in the high pressure ratio range.

The increased pressure ratio and compressor discharge temperature also have a significant influence on the COP of the HACHP. As may be seen the COP contours resemble those of the pressure ratio and compressor discharge temperature. The COP is greatly reduced for the combinations with high pressure ratios as the increased compressor discharge temperature increases the entropy generation in the gas cooler and mixer due to the very large temperature differences. At high ammonia mass fraction and circulation ratios the COP decreases again when exiting the area of high pressure ratio. This is due to increased entropy generation in the mixing. For all ammonia mass fractions above

0.3 one circulation ratio exists that optimizes the COP. As may be seen, the higher the ammonia mass fraction, the lower the optimal circulation ratio.

As the VHC is mainly influenced by the specific volume in the suction line the low pressure governs this parameter. This is apparent when comparing the contours of the VHC and low pressure. All important design parameters are greatly influenced by both ammonia mass fraction and circulation ratio and thus the behaviour of these cannot be attributed solely to one of the two. Thus the main conclusion derived from Figure 3 is that the correct combination of ammonia mass fraction and circulation ratio is needed in order to identify a feasible design.

3.2 Feasible design combination of x_r and f

In order to evaluate the applicability of the HACHP for high temperature concepts ($T_{sink,out}$>100 °C) the feasible combinations of ammonia mass fraction and circulation rates must be identified in this temperature range. The feasible design combinations are given by the design constraints. As these are specific to the three technologies (standard, high pressure NH_3 and transcritical CO_2) three sets of feasible combinations exist.

These sets of feasible combinations have been sought at four heat supply temperatures $T_{sink,out}$=100 °C, 125 °C ,150 °C and 175 °C. Again the component inputs from Table 1 are kept constant. The following external operating conditions are also kept constant at these values: $\Delta T_{sink,glide}$=20 K , ΔT_{lift}=25 K, $\dot{m}_{sink} = 1$ kg/s and $\dot{m}_{source}$=2 kg/s.

Figure 4 shows the feasible combinations of the three technologies at the heat supply temperatures evaluated. Figure 4 (a) shows this for $T_{sink,out}$=100 °C. As seen a feasible set of combinations exist for all three technologies. The set belonging to the standard refrigeration components is the smallest of the three and is constrained by the high pressure p_H and compressor discharge temperature T_H. The set belonging to the high pressure NH_3 technology is significantly larger than that of the standard components. This set is constrained by p_H, T_H and COP. The largest set is that of the transcritical CO_2 components. This set is constrained by the VHC, T_H and COP. Further it can be noted that the VHC constraint is dominant over the low pressure constraint in the entire range, thus the low pressure constraint is effectively extraneous.

In Figure 4 (b) the heat supply temperature is increased to 125 °C. As may be seen this has the consequence that the high pressure constraint for the standard components moves beyond the compressor discharge temperature constraint deeming the application of the standard refrigeration components infeasible. A small set of feasible combinations exist for the high pressure NH_3 components. This set is constrained by p_H, T_H. The set of feasible combinations for the transcritical CO_2 components is slightly increased as the VHC constraint is moved back.

Figure 4 (c) has a heat supply temperature of 150 °C. Here it may be seen that no combinations satisfy the constraint of T_H <170 °C. Thus neither standard refrigeration nor high pressure NH_3 components are applicable at a heat supply temperature of 150 °C. The VHC constraint is no longer present due to the high temperature of the source and consequently high desorber pressure. This again causes the set of feasible combinations for transcritical CO_2 components to expand. This set is here only constrained by T_H and COP.

Figure 4 (d) has a heat supply temperature of 175 °C. Here for the first time high pressure constraint for transcritical CO_2, p_H<140 bar, is present. The set of the feasible combinations is reduced compared to a heat supply temperature of 150 °C. The set is constrained by T_H, COP and p_H.

It should be noted that the increased heat supply temperature attained by the use of transcritical CO_2 components is only possible because of the increase tolerance on the compressor discharge temperature. As seen in Figure 4 (c) and (d) a design that keeps the compressor discharge temperature below 170 °C is not possible. Hence if the transcritical CO_2 compressor cannot be modified to sustain these temperatures, these components cannot attain higher supply temperatures than the high pressure NH_3 components. Further it may be seen that the if the standard refrigeration and high pressure NH_3 compressor are modified to sustain 250 °C compressor discharge temperature these can also attain heat supply temperatures up to 175 °C although the set of feasible combinations is considerably smaller. The use of high pressure NH_3 and transcritical CO_2 components with a compressor discharge temperature constraint of 170 °C could still be advantageous as the increased number of feasible solutions could lead to designs with higher efficiency, low investment cost or even both. To assess this thermo-economic analysis may be performed (Bejan *et al.*, 1996).

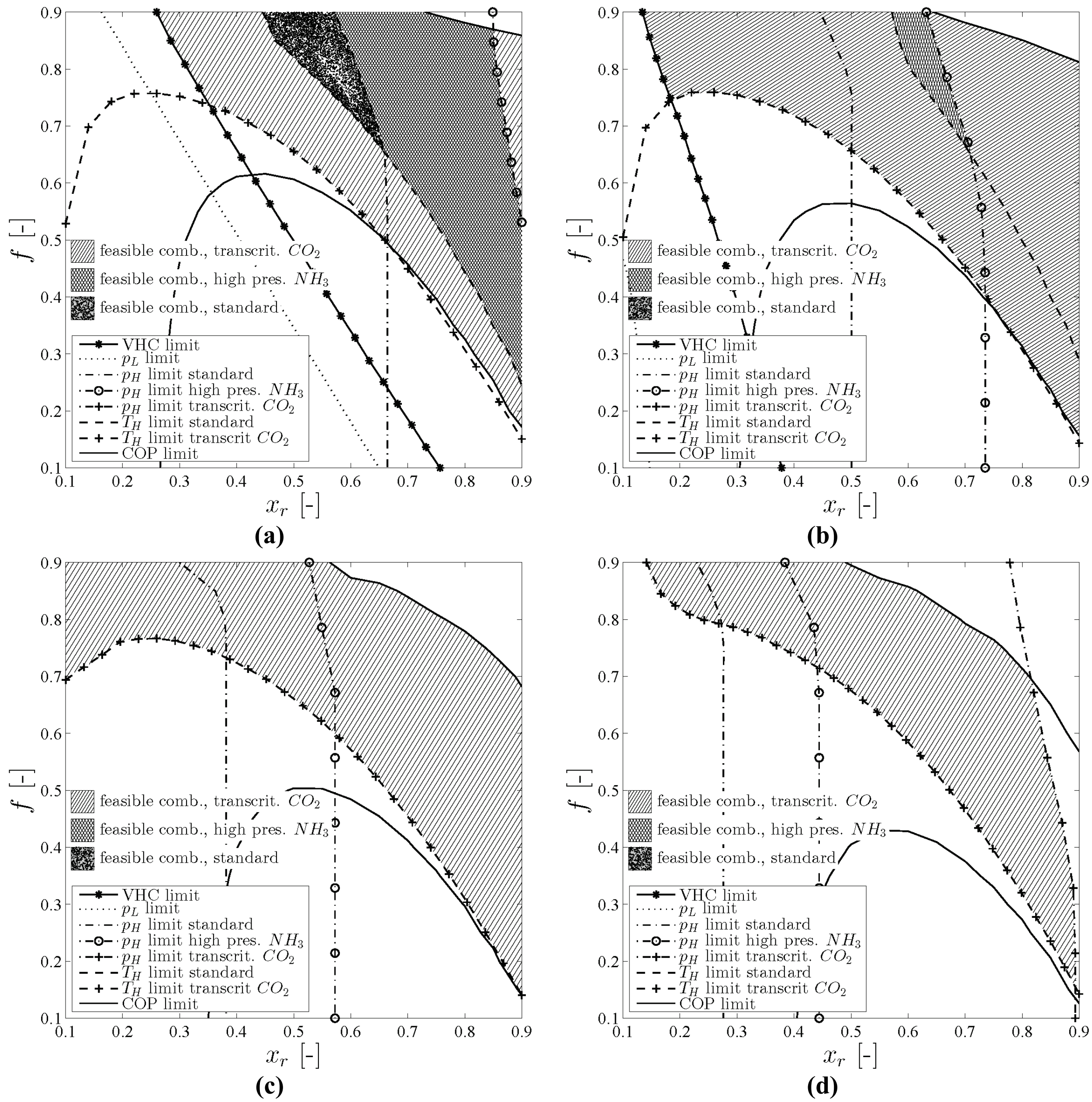

Figure 4. Feasible combination of circulation ratio f and the ammonia mass fraction x_r for standard, high pressure NH_3 and transcritical CO_2 components. Sink glide $\Delta T_{sink,glide}$=20 °C, temperature lift $\Delta T_{sink,glide}$=25 °C, heat supply temperature (a): $T_{sink,out}$=100 °C, (b): $T_{sink,out}$=125 °C, (c): $T_{sink,out}$=150 °C, (d): $T_{sink,out}$=175 °C

In general the compressor discharge temperature shows to be the dominating constraint when assessing the applicability for high temperature design. This could be reduced by implementing a two stage compression and oil cooled compressor. Here it is important that the heat removed in the intercooler and compressor cooler is utilized to heat either the lean solution or the heat sink, otherwise this will lead to a reduction of the COP despite of the reduced compressor work.

Nevertheless, the compressor discharge temperature will set the upper limit of the heat supply temperature. This can be concluded only from the second law as this dictates a positive temperature difference in the gas cooler and thus

$T_H > T_{\text{sink,out}}$. Hence, if the constraint of 170 °C on the compressor discharge temperature cannot be increased then maximum heat supply temperature will never be higher than this.

4. DISCUSSION

Brunin *et al.* (1997) evaluated the HACHP at rich ammonia mass fractions of 0.25, 0.35 and 0.45 and a constant concentration difference of 0.10 between the rich and lean solution. This showed that supply temperatures up to 140 °C can be achieved with a high pressure constraint of 20 bar. This however does not include a constraint on the compressor discharge temperature. Ommen *et al.* (2011) evaluated the HACHP at x_r of 0.7 and 0.9 and a constant source glide of 10 K. The high pressure constraint was 52 bar and the compressor discharge temperature constraint was 160 °C. Here the maximum supply temperature was found to be 125 °C. This is in good agreement with results of the present study. From Figure 4 it is seen that if the constraint on the compressor discharge temperature is not considered, as done by Brunin *et al.* (1997), supply temperatures of 175 °C is possible for standard equipment. This again proves the importance of evaluating all combinations x_r and f. Further it is shown that even when considering the compressor discharge temperature constraint the application of high pressure NH_3 and transcritical CO_2 components as well as an evaluation of all combinations of x_r and f, have expanded the HACHP working domains presented by Brunin *et al.* (1997) and Ommen *et al.* (2011) to cover heat supply temperatures of at least 175 °C.

5. CONCLUSIONS

Results show that all constrained design parameters are highly influenced by the choice of rich ammonia mass fraction and circulation ratio. This emphasises the importance of finding a suitable combination. At circulation ratios below 0.5 and rich ammonia mass fractions between 0.2-0.8 a set of combinations induce a substantial increase in PR. Consequently reducing the COP and increasing the compressor discharge temperature. Designing the HACHP within this set of combinations is therefore not recommended.

Three sets of design constraints have been applied to the choice of rich ammonia mass fraction and circulation ratio, at heat supply temperatures of 100 °C, 125 °C, 150 °C and 175 °C. This shows that the HACHP using standard refrigeration components can attain heat supply temperatures of 100 °C. High pressure NH_3 components increase this to 125 °C. The set of feasible combinations for the standard refrigeration components and the high pressure NH_3 components is constrained mainly by the high pressure and the compressor discharge temperature.

Heat supply temperatures of both 150 °C and 175 °C are possible using transcritical CO_2 components modified to sustain compressor discharge temperatures up to 250 °C. If these modifications cannot be made there is no advantage in terms of heat supply temperature when using transcritical CO_2 components compared to high pressure NH_3.

The dominating constraints when evaluating the maximum heat supply temperature are the high pressure and the compressor discharge temperature. If the attainable heat supply temperature of HACHP is to be increased it is not sufficient to only increase allowable pressure, the allowable compressor discharge temperature must also be increased. The use of two stage compression and oil cooled compressors will reduce the compressor discharge temperature and should be evaluated for high temperature HACHP development.

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NOMENCLATURE

COP	coefficient of performance	(–)	**Subscripts**	
f	circulation ratio	(–)	abs	absorber
h	specific enthalpy	(kJ/kg)	comp	compressor
$\dot{m}$	mass flow rate	(kg/s)	des	desorber
p	pressure	(bar)	dis	displacement
PR	pressure ratio p_H/p_L	(–)	e	electrical
q	vapour mass fraction	(–)	gc	gas cooler
$\dot{Q}$	heat load	(kW)	H	high
T	temperature	(°C (difference K))	ihex	internal heat exchanger
v	Specific volume	(m^3/kg)	is	isentropic
$\dot{V}$	volume flow rate	(m^3/s)	l	lean
VHC	volumetric heat capacity	(kJ/m^3)	L	low
$\dot{W}$	work	(kW)	pump	pump
x	ammonia mass fraction	(–)	pp	pinch point
Abbreviations			r	rich
EES	Engineering equation solver		suc	suction line
HACHP	Hybrid absorption compression heat pump		v	vapour
VCHP	Vapour compression heat pump		vol	volumetric
Greek				
Δ	difference			
ε	effectiveness	(–)		
η	efficiency			

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